

A Comparison of Procedures for Estimating the Impact of Nondensable Gases on Steam Condensation Taking into Account the Interaction of Power Engineering Processes in the Primary Circuit of a Nuclear Power Installation

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Abstract—We consider two procedures for evaluating the influence of nondensables on steam condensation taking into account the integral effect from interaction between the water-chemistry and thermal-hydrodynamic processes occurring in the primary coolant circuit of a nuclear power installation (NPI). Results from calculations of the passive safety system of an NPI are given.

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Among the tasks that have to be solved by designers of the primary coolant circuit of a nuclear power installation (NPI) with the aid of modern approaches is substantiating the serviceability of the reactor installation's heat exchangers. This calls for considering the entire assemblage of power engineering processes, including their interaction, which is especially important when advanced designs employing new types of power technology cycles are to be developed.

Such an approach was implemented at the RNTs KI in cooperation with the MPI Chair of Nuclear Power Stations within the framework of substantiating the operational stability of a passive safety system (PSS), one of the promising solutions aimed at improving the operational safety of the primary coolant circuit of NPIs.

By its functional purpose, the PSS must remain serviceable under loss-of-power-supply conditions for a long period of time and without any intervention from outside. This can be attained by designing the PSS loop so that the coolant in it is driven to flow by natural circulation and the required water chemistry is maintained in it in a self-regulating manner. The primary coolant circuits of research test facilities and boiling-water power-generating reactors at present operate with uncorrected water chemistry. An attempt to dope the coolant with gaseous hydrogen or an ammonia solution for suppressing radiolysis results in a higher content of hydrogen appearing in the steam leaving the reactor core. The problem of setting up a self-regulating water

chemistry has not thus far been solved for existing designs of the boiling-water reactors.

The passive safety system (Fig. 1) is essentially a closed loop with naturally circulating coolant. When passing through the steam-generator tubes, the coolant is exposed to a radiation field, which causes hydrogen and oxygen to arise in it under the effect of radiolysis. The loop also witnesses corrosion processes occurring in it, giving rise to molecular hydrogen. This can impair the normal operation of the installation; in particular, a

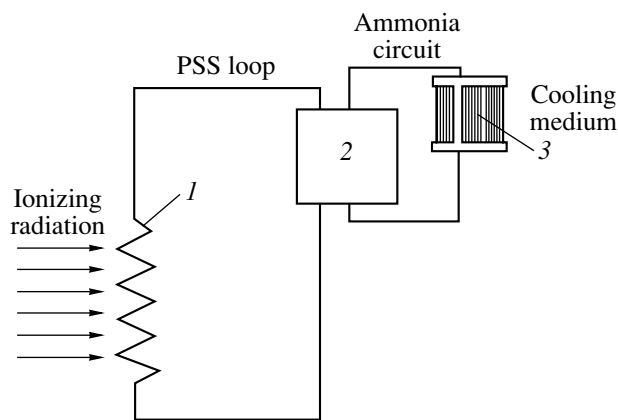


Fig. 1. Basic circuit of the passive safety system. (1) Reactor tubing of the PSS, (2) ammonia heat exchanger, and (3) heat exchanger conveying heat from ammonia to the environment.

drop in the efficiency of condenser operation may upset the circulation of coolant.

The ability of the PSS to operate stably was substantiated by simulating thermal and hydrodynamic processes jointly with water chemistry regimes for different concentrations of excess hydrogen. The following aspects were considered:

- (i) radiation–chemical transformations in the irradiation zones of the economizer and the evaporation sections;
- (ii) passage of radiolysis products over the PSS loop; and
- (iii) interphase mass transfer of the radiolysis products, including nondensables, in the evaporation section and within the condenser.

It was found in the course of simulation that the generation of radiolysis products in the evaporation section is compensated for in a guaranteed manner by their being recombined in the economizer section, which, in turn, results in a stable self-regulating water chemistry. An important aspect of our studies was investigating the interaction between thermal and hydrodynamic conditions and water chemistry, which manifests itself in a buildup of nondensables.

If steam contains nondensable gases, this causes its partial pressure to drop; consequently, the steam condensation temperature drops, which, other factors being equal, results in a smaller heat flux removed during condensation.

The problem of calculating heat transfer during the condensation of steam from a gas–vapor mixture in full statement should be considered as a conjugate one; attempts to find a rigorous analytical solution only meet with success in the simplest cases [1]. Those facing the need to solve such problems usually resort either to difference approximations of the differential equations or to a search for approximate and simpler calculation methods.

This study considers two procedures, one of which was proposed by the authors of this paper (hereinafter referred to as the RNTs KI method); the other is used in RELAP5 (one of the foreign thermohydraulic best-estimate codes) for calculating the effect of nondensables on steam condensation [2]. These procedures were used in conjunction with the appropriate radiation-chemical models developed at the RNTs KI for taking into account the interaction between water chemistry and the thermal-hydrodynamic conditions during the operation of the primary coolant circuits of NPIs.

THE RNTS KI PROCEDURE FOR ESTIMATING THE HEAT TRANSFER DURING THE CONDENSATION OF STEAM FROM A GAS–VAPOR MIXTURE

The main idea of the proposed procedure is to replace the system of equations for the conservation of mass, momentum, and energy of a gas–vapor mixture,

as well as the equation of steam diffusion (the full statement of the problem), with an approximate scheme of heat transfer through a multilayer wall. The effect of nondensable gases on heat transfer during condensation is taken into account by introducing additional heat resistance $R = 1/\tilde{\alpha}_{\text{mix}}$, where $\tilde{\alpha}_{\text{mix}}$ is the effective coefficient for the transfer of heat from the steam–gas core of a flow to the phase interface. Thus, the total heat resistance increases compared with that during the condensation of pure steam; this results in an increase in the total temperature difference at the same heat flux and, as a consequence, in the mass transfer becoming less intensive.

Below we consider a film condensation of steam from a gas–vapor mixture in the course of joint heat-and-mass transfer processes for a turbulent flow of a gas–vapor mixture inside a tube.

Since condensation of a pure saturated steam renders the coolant isothermal, the steam core of the flow shows no transverse temperature gradient, and its heat resistance is equal to zero. In this case, the total heat resistance is composed of three components: $R_{\infty} = 1/\alpha_{\infty}$ for a cooling medium (α_{∞} is the coefficient of heat transfer in the bulk of a cooling fluid); $R_w = \delta_w/\lambda_w$ for a heat-transferring wall (δ_w and λ_w are the wall thickness and heat conductance, respectively); and $R_l = 1/\alpha_l$ for the turbulent film of liquid (condensate), where α_l is the heat-transfer coefficient of the film).

The heat flux removed from the wall during condensation is given by

$$q = k(T_s - T_{\infty}), \quad T_s = T_{\text{int}}, \quad (1)$$

where k is the heat-transfer coefficient found from the relation

$$\frac{1}{k} = R_{\infty} + R_w + R_l = \frac{1}{\alpha_{\infty}} + \frac{\delta_w}{\lambda_w} + \frac{1}{\alpha_l}, \quad (2)$$

where T_s , T_{int} , and T_{∞} are the temperatures of steam, phase interface, and cooling medium.

The decrease in the steam saturation temperature at the interface of phases $T_{\text{int}} = T_{\text{sat}}(p_{\text{p.int}})$ (where $p_{\text{p.int}}$ is the partial pressure of steam at the phase interface), which is caused by the drop in $p_{\text{p.int}}$ due to the presence of nondensables in steam, leads, in turn, to a reduction in the available difference of condensation temperatures $\Delta T = T_{\text{int}} - T_{\infty}$. The mass flow of steam j removed through the phase interface is written as [3]

$$j = \rho_s \beta \ln(\varepsilon_{\text{int}}/\varepsilon_0), \quad (3)$$

where ρ_s is the density of steam; β is a mass-transfer coefficient; $\varepsilon = p_g/p$ is the molar fraction (dimensionless partial pressure) of the gas component; and the subscripts “0” and “int” stand for the core of a steam–gas flow and the liquid–steam interface, respectively.

It should be borne in mind that the sum of the partial pressures of steam and gas remains constant at any

point of the flow and is equal to the pressure of the gas–vapor mixture,

$$p_g + p_s = p. \quad (4)$$

In most cases, we can assume that the change in the enthalpy of a condensate layer due to its depression is negligibly small. Thus, the temperature of the phase interface (the condensation surface) T_{int} can be found from the following relation [4]:

$$k(T_{\text{int}} - T_{\infty}) = f\alpha_{\text{mix}} \left(\frac{r}{c_{ps}} + \frac{T_0 - T_{\text{int}}}{1 - e^{-f}} \right), \quad (5)$$

where r is the specific heat of vaporization, α_{mix} is the coefficient of heat transfer for a single-phase gas–vapor mixture flowing in a tube (when there is no liquid film), and T_0 is the temperature of the gas–vapor mixture (the flow core).

The dimensionless parameter f is given by

$$f = \frac{c_{ps}}{c_{p \text{ mix}} \text{Le}^{0.6}} \ln \frac{\epsilon_{\text{int}}}{\epsilon_0}, \quad (6)$$

where c_{ps} and $c_{p \text{ mix}}$ are the specific heat of steam and that of steam–gas mixture, respectively; $\text{Le} = D_{\text{mix}}/\alpha_{\text{mix}}$ is the Lewis number; and D_{mix} and α_{mix} are, respectively, the diffusion constant and the thermal diffusivity coefficient of the mixture.

The mass-transfer coefficient is determined from the Reynolds analogy between heat and mass transfer for a flow of a steam–gas mixture,

$$\beta = \alpha_{\text{mix}}/(\rho_{\text{mix}} c_{p \text{ mix}} \text{Le}^{0.6}), \quad (7)$$

where ρ_{mix} is the density of the gas–vapor mixture.

Using (3), (6), and (7), we can derive the following relation for calculating the mass flow of steam:

$$j = \alpha_{\text{mix}} f / c_{ps}. \quad (8)$$

Since a steam–gas mixture contains gas, the steam saturation pressure in the flow core will be lower than the pressure of the mixture: $p_{s0} = p(1 - \epsilon) < p$. So, the steam saturation temperature in the flow core will also be lower than that of the gas–vapor mixture: $T_s = T_{\text{sat}}(p_{s0}) < T_0$ (for superheated steam). Introducing, in accordance with [5], the effective coefficient of heat transfer $\tilde{\alpha}_{\text{mix}}$, we can calculate the heat flux from saturated steam in the flow core to the phase interface using the following formula:

$$q = \tilde{\alpha}_{\text{mix}}(T_s - T_{\text{int}}). \quad (9)$$

Furthermore, if we introduce the effective heat-transfer factor \tilde{k} , we come to the formula

$$q = \tilde{k}(T_s - T_{\infty}). \quad (10)$$

The change from (1) to (10) in writing an expression for heat flux density is equivalent to inserting an additional section into a circuit of series-connected heat

resistances. This causes the external heat-transfer boundary to move some distance away ($T_{\text{int}} \Rightarrow T_s$); that is, it results in the total heat resistance increasing by $R_{\text{mix}} = 1/\tilde{\alpha}_{\text{mix}}$. Thus, the expression for the effective heat-transfer coefficient will become

$$\frac{1}{\tilde{k}} = R_{\infty} + R_w + R_l + R_{\text{mix}} = \frac{1}{\alpha_{\infty}} + \frac{\delta_w}{\lambda_w} + \frac{1}{\alpha_l} + \frac{1}{\tilde{\alpha}_{\text{mix}}}. \quad (11)$$

It should be noted that, although introducing the effective heat-transfer coefficients does not allow the original problem to be solved, it allows light to be thrown on the physical picture of complex interrelated and conjugate processes of heat and mass transfer that occur during condensation in the presence of nondensables. In [5], the results from investigations by German researchers into heat transfer during condensation from a gas–vapor mixture were analyzed. To this end, it was proposed to use the following “influencing parameter,” which determines the nature of heat transfer during condensation:

$$A \equiv k/\tilde{\alpha}_{\text{mix}} = (T_s - T_{\text{int}})/(T_{\text{int}} - T_{\infty}). \quad (12)$$

With $A \rightarrow \infty$, the heat resistance of the flow’s gaseous core has a prevailing effect ($R_{\text{mix}} \rightarrow \infty$). In this case, $\tilde{k} \approx \tilde{\alpha}_{\text{mix}}$ and $T_{\text{int}} \approx T_{\infty}$. Such relations are observed when the molar fraction of gas in the core approaches unity ($\epsilon_0 \rightarrow 1$) and when the coefficient for the transfer of heat from the flow’s gaseous core has the low values ($\alpha_{\text{mix}} \rightarrow 0$). In practice, these conditions are observed to take place at the condenser outlet.

When $A \rightarrow 0$, the transfer of heat is mainly influenced by the total heat resistance as viewed from the condensate end; i.e., $(R_{\infty} + R_w + R_l) \rightarrow \infty$. In this case, $k \approx \tilde{k}$ and $T_s \approx T_{\text{int}}$. Physically, this limiting case takes place at low concentrations of a noncondensing gas ($\epsilon_0 \rightarrow 0$) and when the coefficient for the transfer of heat from the flow’s gaseous core has large values ($\alpha_{\text{mix}} \rightarrow \infty$). In practice, this corresponds to the conditions found at the condenser inlet.

The following boundaries are proposed in [5] to demarcate the regions in which the mechanisms of mass- and heat transfer have a predominating effect: $A > 2.0$ for mass transfer, $A < 0.5$ for heat transfer, and $0.5 \leq A \leq 2.0$ for the intermediate region.

Following the data of [6], we introduce in this study the following relative law of a heat transfer when $T_0 - T_{\infty} = \text{idem}$:

$$\psi = q/q_*, \quad (13)$$

where q_* is the heat flux for the “benchmark” case when the pure steam condenses, which corresponds to the equality of all the three temperatures, i.e., the gas–vapor mixture temperature T_0 , the steam saturation temperature at the specified pressure in the flow $T_{\text{sat}}(p)$,

and the phase interface temperature T_{int} . Using (1) we can present (13) as

$$\psi = (T_{\text{int}} - T_{\infty}) / (T_0 - T_{\infty}) \quad (14)$$

for $T_0 - T_{\infty} = \text{idem}$.

From (14) it follows that $\psi = 1$ at $\epsilon_0 = 0$ and $\psi < 1$ at $\epsilon_0 > 0$. Thus, for the problem of steam condensation from a gas–vapor mixture to be solved in practice, we must determine the effect of nondensable gases on the phase interface temperature T_{int} . This procedure is usually iterative in nature.

As we see from Eqs. (1) and (2), given the known values of the cooling medium temperature T_{∞} and the heat-transfer coefficient k , the problem of evaluating the heat flux removed from the wall during condensation boils down to calculating the steam saturation temperature at the phase interface $T_{\text{int}} = T_{\text{sat}}(p_{\text{p.int}})$.

The dimensionless coefficient of heat transfer during the condensation of pure steam for a turbulent flow of liquid film is calculated from the Boiko–Kruzhilin formula [1]

$$\text{Nu} = 0.024 \text{Re}_c^{0.8} \text{Pr}_l^{0.43} \left(1 + \sqrt{1 + \frac{\rho_l}{\rho_s} x_0} \right), \quad (15)$$

where $\text{Nu} = \alpha d_0 / \lambda_l$ is the Nusselt number; d_0 is the channel diameter; $\text{Pr}_l = \nu_l / a_l$ is the Prandtl number for liquid; a_l , λ_l , and ν_l are the thermal diffusivity, heat conductivity, and kinematic viscosity coefficient of liquid, respectively; $\text{Re}_c = u_c d_0 / \nu_l$ is the Reynolds number found from the circulation velocity $u_c = j / \rho_l = 4G / (\pi d_0^2 \rho_l)$; and $j = 4G / (\pi d_0^2)$ is the mass velocity of a two-phase mixture.

The dimensionless coefficient of heat transfer for a turbulent flow of a single-phase gas–vapor mixture in a tube is found from the Petukhov–Kirillov formula [6]

$$\text{Nu} = \frac{(\xi/8) \text{PrRe}}{K + 12.7 \sqrt{\xi/8} (\text{Pr}^{2/3} - 1)}, \quad (16)$$

where $\xi = (1.82 \log \text{Re} - 1.64)^{-2}$ is the pressure loss, $K = 1 + 900/\text{Re}$, and Re and Pr are the Reynolds and Prandtl numbers found from the parameters of the single-phase gas–vapor flow.

CALCULATION PROCEDURE USED IN THE RELAP5 THERMAL-HYDRAULIC CODE

When a flow contains nondensables, the steam core of the flow becomes as if isolated from the condensate film. The intensity of the condensation and heat-transfer processes depends on the degree to which the wall is subcooled with respect to the saturation temperature (as found from the partial pressure of steam) and on other factors, e.g., the thickness of the condensate film and the degree of turbulence. The RELAP5 code allows the coefficient for the transfer of heat from a flow's

gas–vapor core to the wall to be calculated proceeding from the following assumptions:

(i) the wall temperature is lower than the steam saturation temperature corresponding to its partial pressure minus 0.001°C (this operation is required to ensure that the procedure for determining the phase interface temperature converges);

(ii) the temperature of the condensate liquid film is higher than that of the wall (this implies that condensation will arise when the heat flux has a direction from the film toward the wall);

(iii) the volume fraction of the liquid phase is more than 0.1, since a change to convective heat transfer occurs at low void fractions;

(iv) the molar fraction of gas is below 0.999; and

(v) the pressure is below the critical value.

The effects of some other factors are taken into account for smoothing the values of the physical quantities in the presence of nondensables. For example, when the wall subcooling temperature is less than 1°C , special correlations are applied to obtain a smooth transition between condensation and boiling. Apart from the temperature transition, there is a volumetric transition. When the void fraction is below 0.1, the heat-transfer coefficient approaches a value corresponding to a single-phase convective heat transfer for liquid and to the same for a gas flow when the void fraction is unity.

The method for calculating the coefficient for the transfer of heat during condensation α_c is based on the assumption that steam condenses in a film manner. With α_c known, it becomes possible to determine the heat flux removed from the flow during steam condensation. The full heat flux is given by

$$q = \alpha_c (T_w - T_{\text{sat}}), \quad (17)$$

where T_w is the wall temperature and T_{sat} is the steam saturation temperature corresponding to the partial pressure of steam in the flow core.

RELAP5 being a two-phase nonequilibrium code, it theoretically allows exchange of energy between both the liquid and gas phases and the wall to be simulated. The heat flux to the liquid is given by

$$q_l = \alpha_c (T_w - T_l), \quad (18)$$

where T_l is the temperature of liquid in the flow core.

The heat flux to the gas phase can be found as the difference between the total heat flux and the heat flux to the liquid.

The transfer of heat during steam condensation in the presence of nondensables is simulated for a film condensation of steam in inclined, vertical, or horizontal tubes, on tubes having cross-sections of different shapes, and on tube bundles. We use a physical model for steam condensation in a vertical tube, the so-called “Geometry 101” or “Model 101” (it should be noted that the approach describing condensation in a tube

space relies in much to this model), which implies that the heat flux that goes from condensing steam passes through a layer of nondensables and then is completely absorbed by the condensate film. Basing on this conservation law, we can use an iterative procedure to determine the temperature and pressure at the phase interface surface and, then, the heat flux rate.

The heat transfer evaluation engine built in the RELAP5 code employs the principle of choosing a maximum between the heat-transfer coefficients for laminar and turbulent flows.

The quantity used as a key parameter for vertical surfaces is condensate film thickness δ . The heat-transfer coefficient for a laminar flow is given by the following simple equation:

$$\alpha_{\text{lam}} = \lambda_l / \delta. \quad (19)$$

The thickness of condensate film is found from the formula

$$\delta = 0.9086 \left(\frac{3\mu_l^2 \text{Re}_l}{g\rho_l \Delta\rho} \right)^{1/3}, \quad (20)$$

where μ_l is the dynamic coefficient for the viscosity of liquid phase, ρ_l is the film density, g is the acceleration of gravity, and $\Delta\rho$ is the difference between the densities of liquid and steam phases.

The Reynolds number for a film is given by

$$\text{Re} = \frac{4G_l}{\pi D \mu_l}, \quad (21)$$

where G_l is the mass flow of liquid and D is the inner diameter of the tube.

As regards a turbulent flow, it is proposed to use the Shach relationship

$$\alpha_t = \alpha_2 (1 + 3.8/Z^{0.95}), \quad (22)$$

where $Z = (1/X - 1)^{0.8} p_{\text{lim}}^{0.4}$ (where X is the mass fraction of the gas phase, which is generally equal to the ratio of the sum of the masses of steam and gas to the sum of the masses of steam, gas, and liquid phase, and $p_{\text{lim}} = p/p_{\text{cr}}$, where p_{cr} is the critical pressure of water) and α_2 is a special coefficient given by

$$\alpha_2 = \alpha_1 (1 - X)^{0.8}. \quad (23)$$

The heat-transfer coefficient α_1 is found from the well-known relationship for a flow of a single-phase liquid

$$\alpha_1 = 0.023 \left(\frac{\lambda_l}{D_h} \right) \text{Re}^{0.8}, \quad (24)$$

where $\text{Re} = GD_h/\mu_l$, G is the total flowrate of a gas-vapor mixture, and D_h is the hydraulic diameter.

The resulting coefficient for the transfer of heat during condensation from a gas-vapor mixture is determined as the maximum value between α_l and α_t ,

$$\alpha_c = \max(\alpha_l, \alpha_t). \quad (25)$$

Vaporization heat flux q_v can be related with phase-to-phase mass flow of steam during condensation j_v through the following equality:

$$q_v = j_v r, \quad (26)$$

where r is the specific heat of evaporation.

The mass flow of steam is given by

$$j_v = \beta \rho_s \ln \left(\frac{1 - p_{\text{p.int}}/p}{1 - p_{s0}/p} \right), \quad (27)$$

where ρ_s is the density of saturated steam under partial pressure at the phase interface, p is the total pressure of gas-vapor mixture, $p_{\text{p.int}}$ is the steam saturation pressure at the phase interface, and p_{s0} is the steam saturation pressure in the flow core.

The density of steam is given by

$$\rho_s = (1 - X) \rho_{\text{mix}}. \quad (28)$$

Thus, by substituting (27) into (26), we come to the expression

$$q_v = r \beta \rho_s \ln \left(\frac{1 - p_{\text{p.int}}/p}{1 - p_{s0}/p} \right). \quad (29)$$

The mass transfer coefficient in (29) is determined as the maximum between the respective values of the coefficient found for laminar and turbulent flows.

The dependence used for a turbulent flow has the form

$$\text{Sh} = 0.023 \text{Re}_g^{0.83} \text{Sc}^{0.44}, \quad (30)$$

where $\text{Sh} = \beta D_h / D_{\text{mix}}$ is the Sherwood number, $\text{Re}_g = \frac{u \rho D_h}{\mu}$ is the Reynolds number for gas, u is the gas phase velocity, $\text{Sc} = \mu_s / \rho_s D_{\text{mix}}$ is the Schmidt number, and μ_s is the dynamic coefficient for steam viscosity.

For a laminar flow, the following holds:

$$\beta D_h / D_{\text{mix}} = 4.0. \quad (31)$$

The heat flux during the condensation of steam from a gas-vapor mixture is given by

$$q_v = \alpha_c (T_{\text{int}} - T_w). \quad (32)$$

Thus, the phase interface temperature must be known if we wish to calculate either the heat flux or the phase-to-phase mass flow of steam.

From (29) and (32) we can obtain the following equation:

$$\alpha_c (T_{\text{int}} - T_w) = r \beta \rho_s \ln \left(\frac{1 - p_{\text{p.int}}/p}{1 - p_{s0}/p} \right). \quad (33)$$

Equation (33) is solved in T_{int} by way of iteration. The value of the wall temperature is used as a first approximation.

Values of parameters in the inlet section of the condenser

Parameter	Excess hydrogen, cm ³				
	0	100	500	1000	5000
Flowrate, kg/s	0.1232	0.1233	0.1236	0.1240	0.1267
Pressure, MPa	3.5700	3.5760	3.5930	3.6140	3.7830
Void fraction	0.7166	0.7193	0.7298	0.7426	0.8435
Steam quality $\times 10^3$	0.68	1.09	2.70	4.69	19.66
Heat-transfer coefficient, W/(m ² K), calculated using:					
RELAP5	6083.8	6096.7	6142.9	6199.9	6638.4
RNTs KI	6387.0	6411.0	6506.0	6633.0	7201.0

ESTIMATING THE INFLUENCE OF NONDENSABLES ON HEAT TRANSFER DURING STEAM CONDENSATION AS APPLIED TO THE NPS PASSIVE SAFETY SYSTEM

The main sources that account for gases entering into the PSS loop are radiolysis of water and corrosion processes. The numerical investigation we had carried out into the PSS water chemistry allowed us to estimate the distribution of radiolysis gases over the PSS loop taking into account the radiolysis processes of accumulation and recombination. The hydrogen liberated as a result of corrosion processes was taken into account in the analysis by introducing an initial excess of hydrogen into the loop.

The main subject of our research was an ammonia heat exchanger, in which steam of the intermediate circuit was condensed. The steam coming into the heat exchanger serves as a heat carrier of the intermediate circuit and carries in it an admixture of nondensables. Calculations have shown that these latter tend to accumulate in the circuit, thus creating a high level of concentration in the upper (inlet) part of the heat exchanger. It is precisely in the inlet part of the heat exchanger that the products of radiolysis will primarily

affect the condensation of steam, a section in which the concentration of nondensables is a maximum and the annular (film-type) mode of condensation arises.

Alternative calculations of the PSS loop were carried out for various initial doses of excess hydrogen in it to estimate the effect on heat transfer of nondensables. From data on the operating conditions, we took the typical levels of initial excess of hydrogen at standard conditions of 0, 100, 500, 1000, and 5000 cm³ and carried out the joint water-chemistry and thermal-hydrodynamic calculations of the loop for each of the above values of initial excess of hydrogen. The purpose of this analysis was to study the effect the excess of hydrogen has on the entire PSS loop. In doing so, we put each initial value of excess hydrogen in correspondence to a unique set of steady thermal and physical parameters of the loop (flowrate, pressure, concentration of nondensables, etc.). Under these conditions we carried out alternative calculations to reveal the effect the initial excess of hydrogen has on the intensity of steam condensation in the ammonia heat exchanger. The effect of nondensables on the transfer of heat during steam condensation in the condenser was evaluated using the two procedures described above.

Some results from the calculations carried out using both the procedures are given in the table and shown in Fig. 2.

Thus, the procedures considered above are simple, efficient, and allow different parameters characterizing the condensation of steam from a gas-vapor mixture to be estimated with acceptable accuracy without resorting to the comparatively complex numerical methods of solving systems of nonlinear and differential equations, which, by the way, are quite demanding in terms of computation time. It should also be mentioned that the calculations carried out using both the procedures give fairly close results.

It can be seen from the calculation results presented above that the steady values of the thermophysical parameters observed at the condenser input will be somewhat different for different initial values of excess hydrogen. This will cause changes to occur in the physical conditions of steam condensation and, consequently, in its intensity. For instance, increasing the

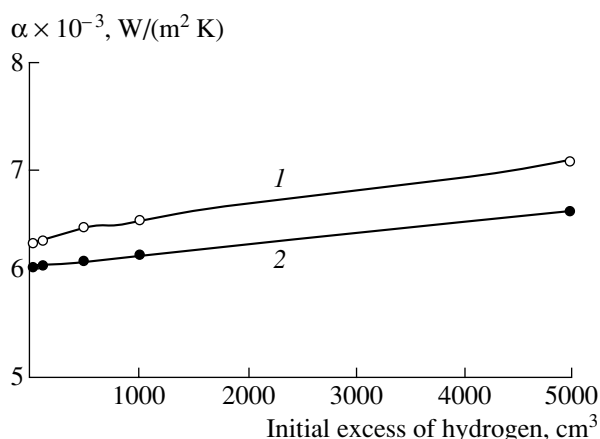


Fig. 2. Coefficient for steady heat transfer as a function of the initial excess of hydrogen. Procedure: (1) RNTs KI; (2) RELAP5.

void fraction at the condenser inlet has the following effect on heat transfer: if we consider a film condensation of steam (Fig. 3), it is exactly the heat resistance of condensate liquid film that constitutes essential part of the heat resistance seen by the process (and actually the only part of this resistance if there is pure steam). In connection with this, an increase in steam fraction in the film flow section is equivalent to a proportional reduction in the thickness of liquid film on the heat-transfer surface and, consequently, to an enhancement of heat transfer during condensation. Parametrically, this can be seen, in particular, from expressions (15), (22), and (23) (these are used both in the RNTs KI procedure and that employed in RELAP5 for calculating the heat resistance of a condensate film), which involve void fraction, a quantity directly influencing the coefficient of heat transfer during steam condensation. Thus, the effect of heat transfer reduction due to a growth of gas content in the heat exchanger is in this case smoothed down by the integrated effects in the loop, primarily, by an increase in the void fraction at the inlet to the apparatus.

Thus, insofar as the PSS loop is concerned, a certain growth in the intensity of condensation (by 10–15%) is observed as the initial excess of hydrogen increases in the circuit; in fact, this is within the accuracy of calculations carried out in accordance with the procedures for evaluating heat transfer for the condensation of steam from a gas–vapor mixture. Nonetheless, the obtained results enable us to conclude that a small positive dynamics in the enhancement of heat transfer during the condensation of steam in an ammonia heat exchanger is possible with an increase in the initial excess of hydrogen within the above range of $0\text{--}5 \times 10^3$ ml (at the standard conditions). The joint calculations of water chemistry and thermal hydrodynamics in the PSS loop allow us to draw the conclusion that a steady water chemistry can settle in the loop, taking into account the generation, accumulation, and recombination of radiolysis and corrosive gases, as well as their effect on the heat-and-mass transfer processes in the loop.

Unlike the majority of similar calculations, the model we used to describe the influence of nondensables on steam condensation took into account the combined integral effects of the interaction between water-chemistry and thermal-hydrodynamic processes in the loop. In our analysis of the effect of nondensables on steam condensation in an ammonia heat exchanger, the latter was not “disjointed” from the loop. The process was investigated taking into account the effect of feedbacks in the loop that cause changes to occur in the other thermophysical properties of coolant at the inlet to the apparatus, rather than by merely varying the inlet concentrations of gases. Such an approach allowed us to obtain a more comprehensive picture of the processes taking place in the PSS loop and to obtain results

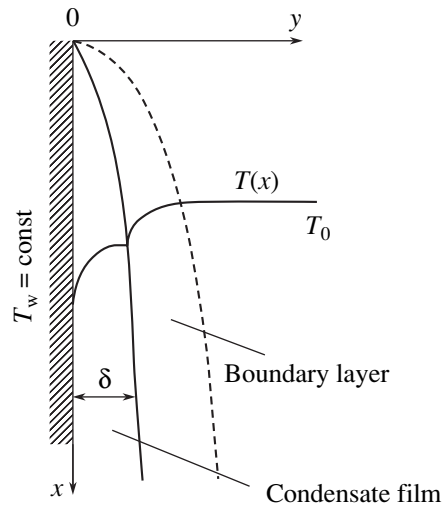


Fig. 3. Scheme of film-type steam condensation.

fundamentally different from those obtained without regard for the effect of feedbacks. An important aspect of this study is that we have substantiated the possibility of obtaining steady self-regulating water chemistry taking into account the effect of nondensables on the processes of heat- and mass transfer in a closed loop.

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